Near-Field Levitation Generated by Ultrasonic Vibrations : Theoretical Analysis and Experiments

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Introduction

With a view to obtaining specific displacement conditions, such as friction cancellation, higher speed or non contact transportation of planar objects, film air bearings using piezoelectric bending elements [1] have many advantages in comparison with conventional air bearing systems, because they are autonomous and they take a reduced place.

To generate an air bearing, a high-pressure difference is created between the external pressure and the pressure in the fluid layer. Film air bearings using a piezoelectric bimorph are performed by a vibration of either the wearing surface [2], or the object to be lifted.

In this paper, after the explanation of the working principle, several prototypes, which have been designed using the ATILA finite element code and built [3] are described. The experimental procedure is detailed with a view to measuring the air bearing thickness and its relative variations. It relies upon the measurement of the air bearing capacitance, using amplitude modulation. Adding several masses on the bimorph, the variations of the air bearing thickness and its relative variations are studied. Then, theoretical model, based on the acoustic radiation pressure [4], is developed to evaluate the mean acoustic pressure in the fluid. This approach is in good qualitative agreement with experiments. Comparing experimental results with numerical results allows to understand physical behavior of the system and gives the way for new experiments.

Working principle

In the considered system, the film air bearing is obtained by prescribing vibrations on the object to be lifted. Therefore, a piezoelectric bimorph (Fig. 1), made of one passive disk stuck on one piezoelectric disk is used. The system is excited at the frequency of its first bending mode. The nodal vibration line is used for the connection system, thus all the vibration energy is confined in the element. For a sufficiently high vibration amplitude, an air bearing appears between the bimorph and the ground.

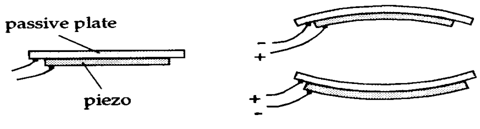

Figure 1: Bending motion of the piezoelectric bimorph

Experiments

Prototypes

To design the prototypes, a finite element analysis with the help of the ATILA code [3] is used. It allows the resonance frequencies and the position of the nodal point, used to fix the system, to be determined. To get a non zero mean volumic displacement on the vibrating face of the prototype, the bimorph transducer is partially clamped by an additional mass at the circumference, which total thickness is 4.5 mm. The thickness at the center of the brass disk is 1.5 mm; the thickness of the PZT is 1 mm. The external radius of the PZT disk is 15 mm, whereas the external radius of the brass is 18.5 mm. The resonance frequency is 13.5 kHz. At the position of the nodal point, a rubber ring allows to add variable masses on the system (Fig. 2).

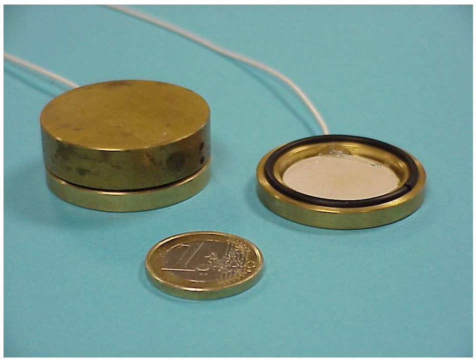

Figure 2. Picture of the prototype.
A rubber ring (right picture) allows variable masses on the system (left picture) to be added

Experimental results

First, resonance and antiresonance frequencies as well as damping are measured by impedancemetry.

The basic circuit used to measure the thickness of the air gap is a capacitive divider, as explained in details in Ref [5].

Several given masses are added on the prototype. Table 1 presents the results for a constant excitation (V= 27 V_{eff}, i = 21 mA_{eff}) and different loadings.

With the mean capacitance, the mean thickness h_0 is deduced: $h_0=\varepsilon_0 S/Ca$, where ε_0 is the dielectric permittivity and S is the surface of the disk. Then, from the amplitude modulation rate $(\Delta V/V)$ and the numerical displacement distribution on the bimorph, the displacement at the center of the bimorph $U(r=0)$ is deduced. One can notice that the modulation rate $(\Delta V/V)$ is very small. The ratio between the

displacement at the center of the bimorph and the mean displacement is around 10. This high value could produce a loss of accuracy in the measurements.
Heavier is the loading, smaller is the mean thickness of the air. The value of the displacement at the center of the bimorph *U(r=0)* is identical (around 12µm), because the excitation current is identical in all the cases, and because there is a mechanical decoupling between the bimorph and the masses. The difference observed for the smaller mass is not yet explained.
The value of the mean thickness h_0 raises the question of the acoustical behaviour of the air layer. In air, the thermal characteristic length is 60 nm. At 13.5 kHz, the thermal boundary-layer thickness (20 µm) is of the same order of magnitude than the thickness of the air gap (Table 1). Therefore, acoustics in the air layer has to be described using an intermediate condition between adiabatic and isothermal conditions [6].

Total mass (g)	Mean Capacitance *Cm*	*ΔV/V* (%)	h_0 (µm)	*U(r=0)* (µm)
26.5	244 pF	2.65	39	9.4
115	338 pF	5.5	28.3	12.7
174	411 pF	6.85	23.4	12.2
188	420 pF	8	22.9	13.2

Table 1. For a constant electrical current excitation and for different masses, mean capacitance *Cm*, amplitude modulation rate *(ΔV/V)*, mean thickness of the air gap displacement h_0, displacement at the center of the bimorph *U(r=0)*.

Theoretical analysis

The study of the acoustic radiation pressure, i.e. the mean excess pressure $<p^E-p_0>$ experienced by a material surface in a sound wave, is necessary to model the problem [4]. If there is no constraint, then $<p^E-p_0> = <V>-<K>$, where $<V> = <p^2>/2\rho c^2$ and $<K> = \rho<u.u>/2$ are the time-averaged potential and kinetic densities. The linear wave equation, using the finite element method, is solved to determine the acoustic fields. Then the Eulerian mean excess pressure is evaluated and the lifted mass is deduced. Fig. 3 presents the variations of the lifted mass as a function of the air layer thickness at 13 kHz, calculated using various values of compressibility in the air layer, to model intermediate conditions between adiabatic and isothermal conditions. The dots correspond to the experiments (Table 1). Numerical approach gives exactly the same trends than experiments. Fig. 3 could be used to determine acoustical properties of the fluid layer. However, numerical calculations are shifted as soon as the frequency is changed. Therefore, a more precise numerical – experimental comparison of the displacement at the bimorph center is required for further analysis. Additional experiments have to be performed with a laser vibrometer to know more precisely the bimorph displacement.

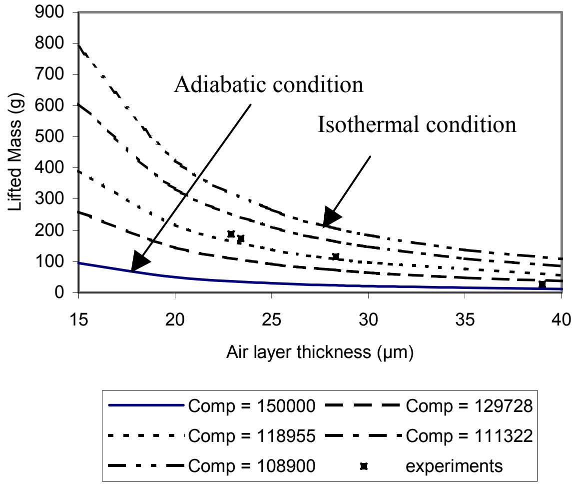

Figure 3. Numerical variations of the lifted mass (g) versus the air layer thickness (µm), calculated for various values of compressibility in the air layer, from adiabatic condition to isothermal condition. Dots correspond to experiments. f = 13kHz.

Conclusion

A film air bearing system has been designed and built. Using the electrical measurement of the capacitance, the mean thickness of the air film and its relative variations have been evaluated. A numerical approach [3,4] using the acoustic radiation pressure, predicts lifted mass in good qualitative agreement with experiments. Additional experimental analysis will be carried out using a laser vibrometer, to measure accurately the mean thickness, its relative variations and the displacement distribution on the bimorph.

References

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